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ADDITIONAL ENERGY LOSSES IN COOLED GAS-TURBINE POWERPLANTS AND THEIR INFLUENCE ON SPECIFIC CYCLE PARAMETERS

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ADDITIONAL ENERGY LOSSES IN COOLED GAS-TURBINE POWERPLANTS AND THEIR INFLUENCE ON SPECIFIC CYCLE PARAMETERS /152*

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ABSTRACT

Discussion of the additional energy losses that take place in a gas turbine with air- or liquid-cooled vanes, as compared to the uncooled version of the turbine. Three types of principal losses are examined: (1) additional hydraulic losses caused by both the inferior geometry of the working section and the structural changes in the cooled boundary layer on the rotor vanes, which, it is shown, for an appropriate geometry of the cooled vanes need not exceed 1.5 to 2 percent; (2) additional thermodynamic losses arising from heat release by the gas when expanding in the working section, which, in the case of air-cooling, are shown to be small but, in the case of multistage high-temperature turbines employing liquid cooling, will exceed the combined other losses; (3) additional energy losses associated with the supply of the cooling medium to the vanes, which are low in the case of liquid-cooling but constitute the major part of the total losses for air-cooling and depend primarily on the parameters

*Numbers given in the margin indicate the pagination in the original foreign text.

and relative flow rate of the coolant. The effect of the individual type of losses on the cycle parameters is examined.

author

The introduction of cooling in gas-turbine power plants gives rise to energy losses which are greater in comparison to those found in the noncooled variant. These principal energy losses are examined below.

1. The additional hydraulic losses ($\zeta_{a.h}$) appear because of the inevitable flow geometry deterioration, in comparision with the noncooled variant, and the structural alteration in the cooled boundary layer of the actuating blades. Detailed examination of these losses shows that a rational choice of the cooling grate geometry (with the compromise between a high efficiency requirement and a high cooling effectiveness) may give $\zeta_{a.h}$ values not exceeding 0.015 - 0.02.

2. The additional thermodynamic losses ($\Delta\bar{H}_0$) appear because of the heat removal from the gas during the process of its expansion in the turbine proper. In the air-cooled turbines these losses are usually small (ref. 1).

In high temperature multistage turbines with the intense liquid cooling they constitute the main portion of the additional losses, which can amount to 5-6 percent (refs. 2 and 3) of the available turbine heat gradient (H_0).

As a result of an assumption of uniform turbine heat gradient distribution, an approximate (accurate to within 10 percent) formula for the determination of $\Delta\bar{H}_0$ was obtained

$$\Delta\bar{H}_0 = \frac{\Delta H_0}{H_0} \approx \bar{q}_c \cdot \delta \left(1 - \frac{1-\delta}{\frac{4z}{a_c} - \delta}\right). \qquad (1)$$

where $\bar{q}_c$ is heat released due to cooling in the region of H_0.

A quantitative estimate of $\bar{q}_c$ may be made according to references 1, 4 $\delta = 1 - \frac{1}{\delta_t^m}$ (δ_t - degree of turbine pressure reduction), $m = \frac{(k - 1)}{k}$ (k - isoenthropy indicator), z - number of stages, n_c - number of cooling banks. The approximate relationship is

$$n_c \approx 2 \frac{c_p T_a}{h_s} \tau (1 - \bar{T}_{al}), \qquad (2)$$

where c_p - specific heat, T_a - atmospheric air temperature, h_s - stage heat gradient, $\tau = \frac{T^*_3}{T_a}$ - cycle temperature ratio increase, $\bar{T}_{al} = \frac{T_{al}}{T^*_3}$ (T_{al} - allowable blade temperature). /153

Taking into account the additional hydraulic and thermodynamic losses due to cooling, the relative internal turbine efficiency is

$$\eta_{t.c.} = \eta_t (1 - \zeta_{a.h}) (1 - \Delta \bar{H}_0) = \eta'_t (1 - \Delta \bar{H}_0). \qquad (3)$$

where η_t - relative internal noncooled turbine efficiency including the heat recovery ratio, η'_t - as above with consideration of the additional hydraulic losses.

3. Additional energy losses associated with the selection, preparation and the delivery of the coolant to components arise under any method of cooling. With liquid cooling these losses are usually small. On the contrary, with air-cooling they constitute the principal portion of the additional losses and they depend mainly on the coolant parameters and its relative consumption ($\bar{G}_c$).

Obviously, the amount of gas entering the cooling stage will be

$$\bar{G}_g = \frac{G_g}{G_B} = (1 - \bar{G}_c) k_\alpha , \qquad (4)$$

where G_B is the quantity of air passing through the first compressor stage and $K_\alpha = \frac{1 + \alpha l_0}{\alpha l_0} \approx 1.015 - 1.025$.

The magnitude of $\bar{G}_c$ may be evaluated according to reference 1.

With the aid of formulas (1)-(4) an analysis may be made of the effect of losses due to cooling on the specific cycle parameters of various types of gas-turbine power plants and on the optimum choice of $\pi_{k.opt}$.

Thus, for the cooled nonairborne gas-turbine power plant with heat regeneration of the withdrawing gases (discounting the effect of the cooling air in the turbine subsequently to its delivery into the turbine proper) we have

$$L_{e_{GTPPc}} = c_p T_a \left(\tau \eta_{T\,c} \quad \bar{G}_g \delta - \frac{1}{\eta_K} \frac{\Pi}{1 - \Pi}\right) \tag{5}$$

$$\eta_{e_{GTPPc}} = \eta_m \eta_g \frac{\delta \eta_{T.c} \quad - \frac{1}{\tau \eta_K \bar{G}_g} \frac{\Pi}{1-\Pi}}{k_{td} \cdot \sigma \cdot \eta_{T.c} \quad \cdot \delta + (1-\sigma)\left[k_{td} - \frac{1}{\tau}\left(1 + \frac{1}{\eta_K} \frac{\Pi}{1-\Pi}\right)\right]}, \tag{6}$$

where $\Pi = 1 - \frac{1}{\pi_k^m}$, η_k - isoenthropic compressor efficiency, π_k - degree of gas-turbine power plant compressor pressure increase, η_m, η_g - mechanical efficiency and combustion chamber efficiency, σ - degree of regeneration, $k_{td} = 0.99 - 0.98$ - a factor accounting for the gas temperature decrease due to cool air mixing, δ and Π are related by the expression $\delta = 1 - \frac{1 - \Pi}{\nu^m}$, $\nu = \nu_{ip\ell} \cdot \nu_{op\ell} \cdot \nu_{cp\ell}$ ($\nu_{ip\ell}$ - total inlet pressure loss factor, ν_{opl} - total outlet pressure loss factor, ν_{cpl} - total combustion chamber pressure loss factor). /154

With the aid of the auxiliary Lagrange multipliers from the expressions

$$\begin{gathered}\frac{\partial f_1(L_e)}{\partial \eta_{T.c}} = 0, \quad \frac{\partial f_1(L_e)}{\partial \bar{G}_g} = 0, \quad \frac{\partial f_1(L_e)}{\partial \Pi} = 0, \\ \frac{\partial f_2(\eta_e)}{\partial \eta_{T.c}} = 0, \quad \frac{\partial f_2(\eta_e)}{\partial \bar{G}_g} = 0, \quad \frac{\partial f_2(\eta_e)}{\partial \Pi} = 0\end{gathered} \tag{7}$$

it is easy to find the equations determining $\pi_{k.opt.c}^{Le}$ and $\pi_{k.opt.c}^{\eta e}$. For $\pi_{k.opt.c}^{Lemax}$: we obtain

$$F_1 = 1 - \frac{\nu^m}{a_0 \bar{G}_g (1-\Pi)^2} - (\nu^m - 1 + \Pi)(1+\theta)\bar{q}_c \frac{\eta_T}{\eta_{T.c}} - \\ - \frac{(\nu^m - 1 + \Pi)\bar{G}_c^2}{(1-\bar{G}_c)\tau\eta_K k_{h.g} S (1-\Pi)^2} = 0 \qquad (8)$$

($a_0 = \tau\eta_k\eta_{t.c}$; S - factor determined according to reference 1, $k_{h.g} \approx 1.05 - 1.15$ - factor accounting for the heat exchange through the blade bases).

With $\pi_{k.opt.c}^{\eta emax.}$ we have the equation

$$L_{e_{GTPP_c}} \cdot \frac{\left(\frac{\bar{G}_c}{1-\bar{G}_c}\right)^2}{k_\alpha k_{h.g} S c_p T_a \tau a_0 (1-\Pi)^2} - \\ - k_{td}\sigma\eta_{e_{GTPPmax}}\left[1 - \frac{\eta_T'}{\eta_{T.c}}\bar{q}_c (1+\theta)(\nu^m - 1 + \Pi) - \\ - \frac{1-\sigma}{\sigma}\frac{\nu^m}{k_{td} a_0 (1-\Pi)^2}\right] + F_1 = 0. \qquad (9)$$

With liquid cooling in formulas (5) (6) and (8), (9) it is necessary to assume

$$\bar{G}_g = k_\alpha \text{ and } \bar{G}_c = 0$$

In the analysis of nonregenerative gas-turbine power plants with cooling in (6) and (9) it is necessary to set $\sigma = 0$ and $\nu = \nu_{cp\ell}$.

It is easily seen that for the noncooled case, when $G_c = 0$, $\bar{q}_c = 0$, the regular formulas for the noncooled gas-turbine power plants can be determined from (5), (6), (8), (9).

For the aircraft gas-turbine power plant, as an example, let us present a relationship derived by the indicated method for the cooled high pressure turbounit.

The equivalent specific power

$$N_{e.s.c} = \eta_m \cdot c_p T_a \tau \bar{G}_g \eta_{T.c} \left(1 - \frac{1-\Pi}{\beta} - \frac{\gamma}{a_0 \bar{G}_g} \cdot \frac{\Pi}{1-\Pi}\right) + + \varepsilon \sqrt{\left(1 - \frac{1-\Pi}{\beta}\right) \frac{\eta_{T.c}}{\eta_T'}} - \frac{V^2}{\eta_p}. \tag{10}$$

The equivalent specific fuel consumption

$$C_{e.s.c} \approx \frac{1}{(\alpha l_0 \cdot N_{e.s.})c}. \tag{11}$$

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$$\beta = (\nu_{cp\ell} \cdot \pi_{pi})^m, \quad \gamma = 1 + \frac{\pi_{pi}^m - 1}{\eta_i},$$

$$\varepsilon = \frac{\varphi_{jsf}\, k_\alpha V}{\eta_p} \sqrt{2 c_p T_a \tau \left(1 - \frac{\eta_T'}{\eta_T^*}\right)(1 + \Delta \bar{H}_r)},$$

V - flight speed, η_p - propeller efficiency, $\pi_{p\ i}$ - degree of pressure increase in the input facility, φ_{jsf} - jet speed factor, $\Delta\bar{H}_r$ - heat recovery factor, η_i - efficiency of input facility

$$\alpha l_0 \approx \frac{\omega}{\bar{G}_g\left[\tau - \gamma\left(1 + \frac{1}{\eta_i} \frac{\Pi}{1-\Pi}\right)\right]}, \quad \omega = \frac{H_i \cdot \eta_{cp\ell}}{c_p T_a},$$

H^i - effective calorific value of the fuel.

$\pi_{k.opt.c}^{Nemax.}$ can be found from the equation

$$F_1 = 1 - \frac{\beta\gamma}{a_0 \bar{G}_g (1-\Pi)^2} + \frac{\varepsilon}{2 b_1 \bar{G}_g \sqrt{\left(1 - \frac{1-\Pi}{\beta}\right) \frac{\eta_{T.c}}{\eta_T'}}} - $$

$$- \beta \bar{q}_c (1 + \theta) \frac{\eta_T'}{\eta_{T.c}} \left(1 - \frac{1-\Pi}{\beta}\right)\left[1 + \frac{\varepsilon}{2 b_1 \bar{G}_g \sqrt{\left(1 - \frac{1-\Pi}{\beta}\right) \frac{\eta_{T.c}}{\eta_T'}}}\right] -$$

$$- \frac{k_\alpha \beta \bar{G}_c^2 \left(1 - \frac{1-\Pi}{\beta}\right)}{k_{h\ g} S \tau \eta_K \bar{G}_g (1-\Pi)^2} = 0, \tag{12}$$

$\pi_{k.opt.c}^{Cemin}$ can be obtained from the relationship

$$\frac{al_0 N_{e.s.c}}{\omega} - \frac{\gamma}{\eta_k (1-\Pi)^2} + N_{e.s.c} \cdot \frac{k_\alpha \cdot \bar{G}_c^2}{k_{hg} S\tau \cdot \bar{G}_g (1-\Pi)^2} + \\ + \frac{b_1}{\beta} \cdot \frac{\eta_{T.c}}{\eta_T'} F_1 = 0, \qquad (13)$$

where $b_1 = c_p T_a \cdot \tau \cdot \eta_t^1 \cdot \eta_m$.

Figure 1 shows the relationships between $L_{eG\Gamma PPc}$, $N_{e.s.c}$, $\eta_{eG\Gamma PPc}$ and $C_{e.s.c}$ as functions of π_k for various values of τ, calculated with the aid of these formulas[1].

Figure 2 shows, as an example, how the losses due to cooling influence the magnitudes of the simple (curves 1, 5, 6) and the regenerated (curve 2) cycles.

Curves 8, 10, 11 represent the maximum performance reduction, and 13, 15 - the maximum efficiency of the simple cycle due to additional losses due to cooling.

It should be noted that every point on curves 8-16 has a corresponding optimum degree of compressor pressure increase (by equations (8), (9), (12) and (13)). /157

Other types of aircraft (turbo-jet, turbo-jet with afterburners) and non-airborne gas-turbine power plants (with complex cycle) were analyzed with the same method.

In all cases:

(1) the increase of T_3^* substantially improves the gas-turbine power plant specific cycle parameters,

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[1]The calculations were made: for the high-pressure turbounit with m = 0.25, η_k = 0.85, η_T = 0.9 for the gas-turbine power plant with A (air) m = 0.25, η_k = 0.85, η_T = 0.9 with D (liquid) same, η_T = 0.85.

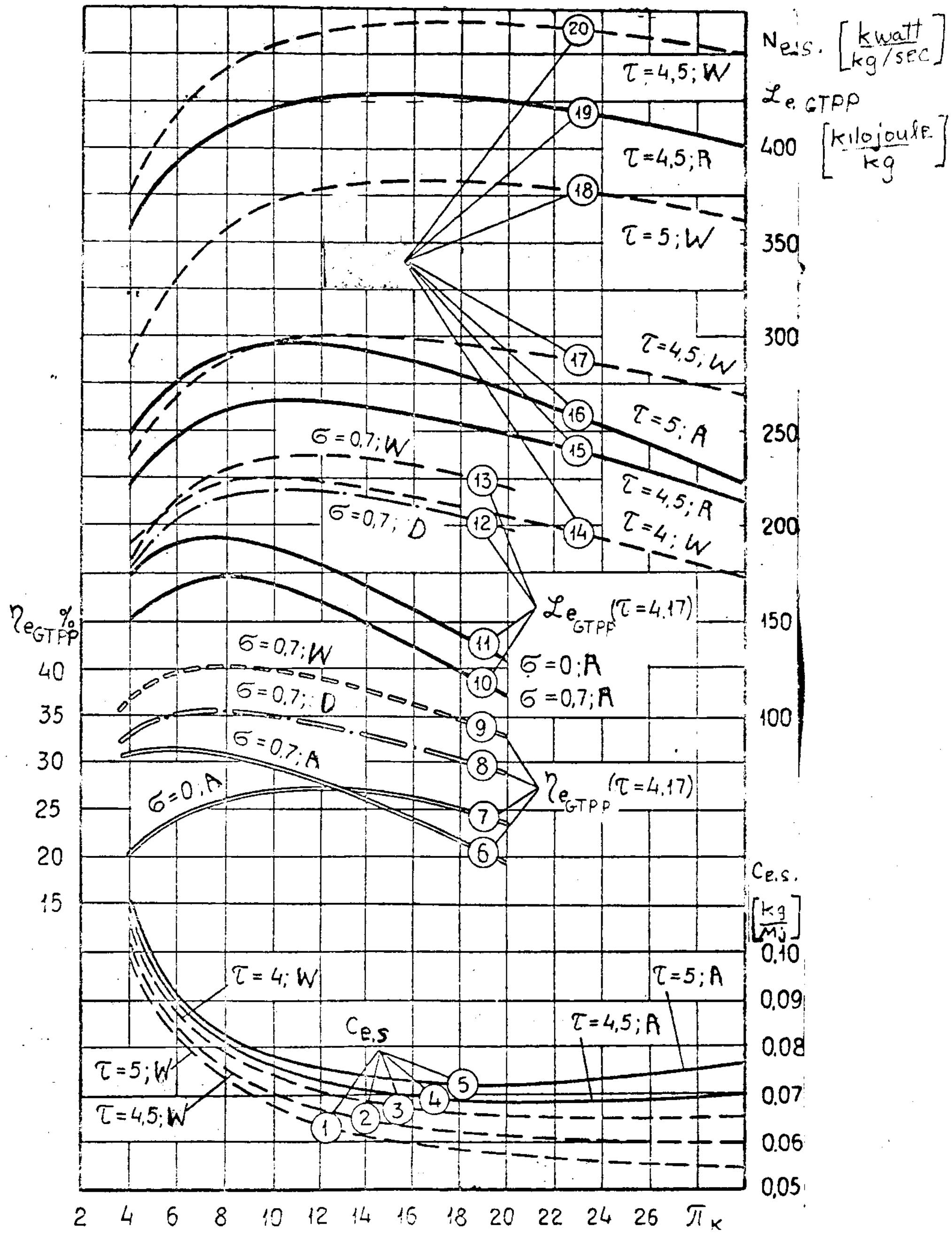

Figure 1. The variation of L_{eGTPP}, η_{eGTPP}, $N_{e.s}$, $C_{e.s}$ as a function of degree of compressor pressure increase (W - without cooling, A - air cooling, D - liquid-cooling) 1 - 5 - $C_{e.s}$ (T_{a1} = 1100°K); 6 - 9 - η_{eGTPP} (T_{a1} = 923°K); 10 - 13 - L_{eGTPP} (T_{a1} = 923°K) 14 - 18 - $N_{e.s}$ (T_{a1} = 1100°K, H = 0 M_H = 0); 19 - 20 - $N_{e.s}$(T_{a1} = 1100°K, H = 11km, M_H= 0.8).

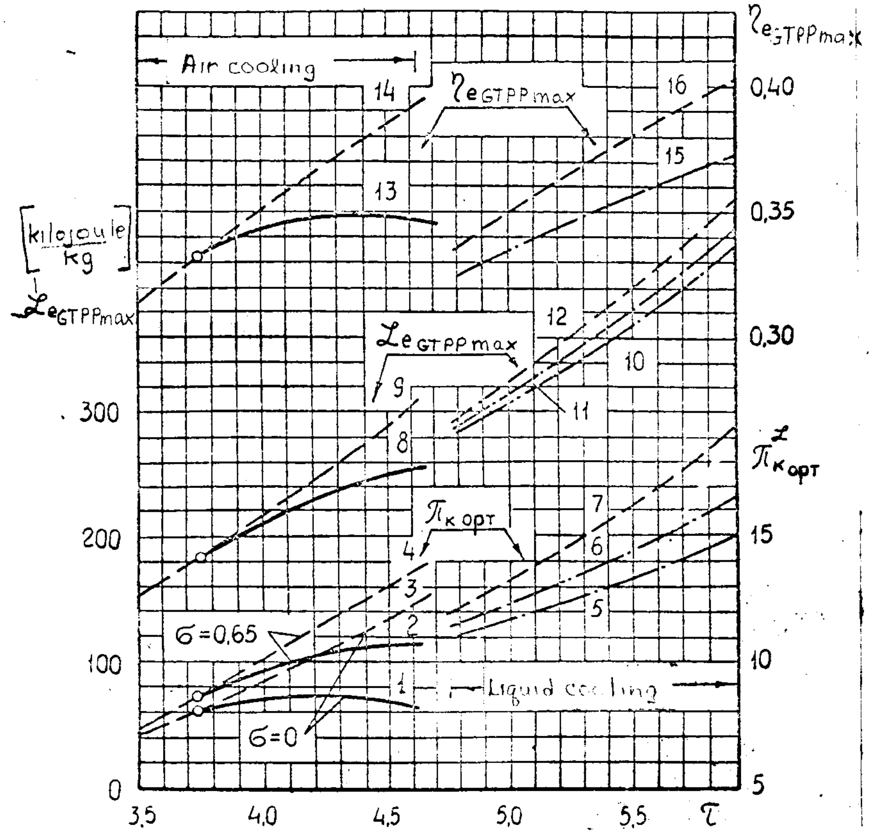

Figure 2. The influence of cooling on the gas-turbine power plant specific indicators and the optimum degree of compressor pressure increase: 1 - 6 - variation of $\pi^{L}_{K.opt}$ (1, 2 with $T_{a1} = 923^{o}K$, 6 with $T_{a1} = 1000^{o}K$), 8 - 12 - variation of $L_{eGTPP\ max}$ (with the same T_{a1}), 13-16 - variation of $\eta_{eGTPPmax}$ (with the same T_{a1}), ------- the non-cooled case.

(2) the additional losses due to cooling noticably diminish the effect of T^{*}_{3} increase, especially when air cooling is used,

(3) the optimum degrees of compressor pressure increase (for the generation of $L_{eGTPPmax}$ or $\eta_{eGTPPmax}$) for the cooled gas-turbine power plant, with the same τ, appear to be substantially lower, than for the noncooled case (compare figs. 1 and 3; 2 and 4; and 5, 6 and 7). The latter indubitably simplifies the problem of creation of a compressor with a high effectiveness.

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